APLIKASI TERMODINAMIKA PADA PROSES ALIR (lanjutan)

The discipline

- Principles: “Fluid mechanics” and “Thermodynamics”
- Contrast
  - Flow process inevitably result from pressure gradients within the fluid. Moreover, temperature, velocity, and even concentration gradients may exist within the flowing fluid.
  - Uniform conditions that prevail at equilibrium in closed system.
- Local state
  - An equation of state applied locally and instantaneously at any point in a fluid system, and that one may invoke a concept of local state, independent of the concept of equilibrium.

FLUIDA

Materi yang bisa mengalir, gas atau cairan

Akspansivitas volume $$\beta = \frac{1}{V}\left(\frac{\partial V}{\partial T}\right)_P$$

Kompresibilitas Isotermal $$\kappa = -\frac{1}{V}\left(\frac{\partial V}{\partial P}\right)_T$$

Fluida kompresibel : volume berubah karena tekanan atau perubahan suhu; mis. gas $\beta \neq 0; \quad \kappa \neq 0$

Fluida inkompresibel : volume **tidak** berubah karena tekanan atau perubahan suhu; suatu idealisasi ; cairan hampir bisa dianggap sebagai fluida inkompresibel. $\beta = 0; \quad \kappa = 0$

Duct flow of compressible fluids

- Equations interrelate the changes occurring in pressure, velocity, cross-sectional area, enthalpy, entropy, and specific volume of the flowing system.
- Consider a adiabatic, steady-state, one dimensional flow of a compressible fluid:

$$\Delta H + \frac{\Delta u^2}{2} = 0 \longrightarrow dH = -udu$$

- The continuity equation: $d(uA/V) = 0 \longrightarrow \frac{dV}{V} - \frac{du}{u} - \frac{dA}{A} = 0$

$$dV = \left(\frac{\partial V}{\partial S}\right)_P dS + \left(\frac{\partial V}{\partial P}\right)_S dP$$

$$\left(\frac{\partial V}{\partial P}\right)_S = -\frac{V^2}{c^2}$$

From physics, c is the speed of sound in a fluid

$$\left(\frac{\partial V}{\partial S}\right)_P = \left(\frac{\partial V}{\partial T}\right)_P \left(\frac{\partial T}{\partial S}\right)_P$$

$$\beta \equiv \frac{1}{V}\left(\frac{\partial V}{\partial T}\right)_P \qquad \left(\frac{\partial S}{\partial T}\right)_P = \frac{C_P}{T}$$

$$\left(\frac{\partial V}{\partial S}\right)_P = \frac{\beta V T}{C_P}$$

$$\frac{dV}{V} = \frac{\beta T}{C_P} dS - \frac{V}{c^2} dP$$

$$\frac{dV}{V} - \frac{du}{u} - \frac{dA}{A} = 0$$

$$\frac{dV}{V} = \frac{\beta T}{C_P} dS - \frac{V}{c^2} dP$$

$$dH = -udu$$

$$dH = TdS + VdP$$

$$\left(1 - \left(\frac{u}{c}\right)^2\right) VdP + \left(1 + \frac{\beta u^2}{C_P}\right) TdS - \frac{u^2}{A} dA = 0$$

$$\mathbf{M} = \left(\frac{u}{c}\right)$$ The Mach number

$$\left(1 - \mathbf{M}^2\right) VdP + \left(1 + \frac{\beta u^2}{C_P}\right) TdS - \frac{u^2}{A} dA = 0$$

$$dH = TdS + VdP$$

Relates du to dS and dA

$$udu - \left(\frac{\frac{\beta u^2}{C_P} + \mathbf{M}^2}{1 - \mathbf{M}^2}\right) TdS + \left(\frac{1}{1 - \mathbf{M}^2}\right) \frac{u^2}{A} dA = 0$$

Pipe flow

$$udu - \left(\frac{\frac{\beta u^2}{C_P} + \mathbf{M}^2}{1-\mathbf{M}^2} \right) TdS + \left(\frac{1}{1-\mathbf{M}^2} \right) \frac{u^2}{A} dA = 0$$

$$\left(1-\mathbf{M}^2\right) VdP + \left(1 + \frac{\beta u^2}{C_P} \right) TdS - \frac{u^2}{A} dA = 0$$

$$u \frac{du}{dx} = T \left(\frac{\frac{\beta u^2}{C_P} + \mathbf{M}^2}{1-\mathbf{M}^2} \right) \frac{dS}{dx}$$

$$\frac{dP}{dx} = -\frac{T}{V} \left(\frac{1 + \frac{\beta u^2}{C_P}}{1-\mathbf{M}^2} \right) \frac{dS}{dx}$$

For subsonic flow, $\mathbf{M}^2 < 1$, $\boxed{\frac{du}{dx} > 0}$ $\boxed{\frac{dP}{dx} < 0}$, the pressure decreases and the velocity increases in the direction of flow. For subsonic flow, the maximum fluid velocity obtained in a pipe of constant cross section is the speed of sound, and this value is reached at the exit of the pipe.

Consider the steady-state, adiabatic, irreversible flow of an incompressible liquid in a horizontal pipe of constant cross-sectional area. Show that (a) the velocity is constant. (b) the temperature increases in the direction of flow. (c) the pressure decreases in the direction of flow.

Control volume: a finite length of horizontal pipe, with entrance (1) and exit (2)

The continuity equation: $$\frac{u_1 A_1}{V_1} = \frac{u_2 A_2}{V_2}$$

incompressible $V_1 = V_2$

const. cross-sectional area $A_1 = A_2$

→ $u_1 = u_2$

Entropy balance (irreversible): $$S_G = S_2 - S_1 > 0$$

↓ incompressible liquid with heat capacity C

$$S_G = S_2 - S_1 = \int_{T_1}^{T_2} C \frac{dT}{T} > 0$$ → $T_2 > T_1$

Energy balance with ($u_1 = u_2$): $H_1 = H_2$ → $$H_2 - H_1 = \int_{T_1}^{T_2} CdT + V(P_2 - P_1) = 0$$

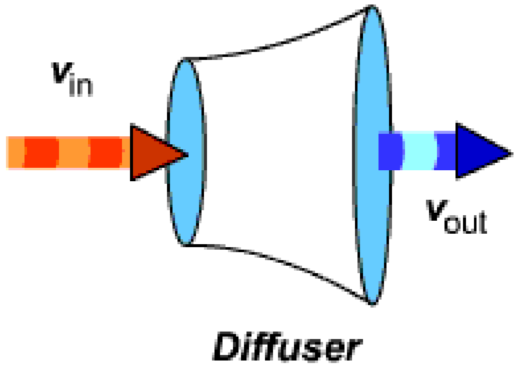
↓ $T_2 > T_1$

$P_2 < P_1$

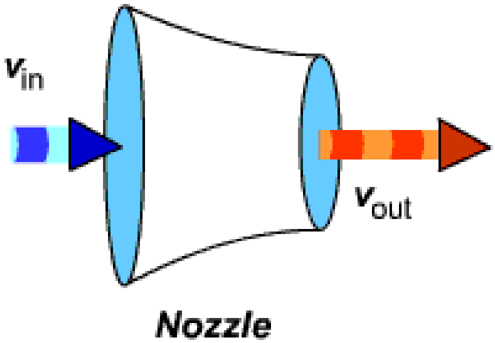
If reversible adiabatic: $T_2 = T_1$; $P_2 = P_1$. The temperature and pressure change originates from flow irreversibility.

Nozzles and Diffusers

A diffuser converts high speed, low pressure flow to low speed, high pressure flow



A nozzle converts high pressure, low speed flow to low pressure, high speed flow



Nozzles:

$$udu - \left(\frac{\frac{\beta u^2}{C_P} + \mathbf{M}^2}{1 - \mathbf{M}^2} \right) TdS + \left(\frac{1}{1-\mathbf{M}^2} \right) \frac{u^2}{A} dA = 0$$

Reversible flow

$$u\frac{du}{dx} + \left(\frac{1}{1-\mathbf{M}^2} \right) \frac{u^2}{A} \frac{dA}{dx} = 0$$

$$\left(1 - \mathbf{M}^2\right) VdP + \left(1 + \frac{\beta u^2}{C_P} \right) TdS - \frac{u^2}{A} dA = 0$$

Reversible flow

$$\left(1 - \mathbf{M}^2\right) V \frac{dP}{dx} - \frac{u^2}{A}\frac{dA}{dx} = 0$$

	Subsonic: $\mathbf{M} < 1$		Supersonic: $\mathbf{M} > 1$	
	Converging	Diverging	Converging	Diverging
$\frac{dA}{dx}$	-	+	-	+
$\frac{dP}{dx}$	-	+	+	-
$\frac{du}{dx}$	+	-	-	+

For subsonic flow in a converging nozzle, the velocity increases as the cross-sectional area diminishes. The maximum value is the speed of sound, reached at the throat.

$$u\frac{du}{dx}+\left(\frac{1}{1-\mathbf{M}^2}\right)\frac{u^2}{A}\frac{dA}{dx}=0$$

$$\left(1-\mathbf{M}^2\right)V\frac{dP}{dx}-\frac{u^2}{A}\frac{dA}{dx}=0$$

isentropic

$$udu=-VdP$$

$$u_2^2-u_1^2=-2\int_{P_1}^{P_2}VdP$$

$$PV^{\gamma}=const.$$

$$u_2^2-u_1^2=\frac{2\gamma P_1V_1}{\gamma-1}\left[1-\left(\frac{P_2}{P_1}\right)^{\gamma-1/\gamma}\right]$$

$$u_2=c$$

$$u_1=0$$

$$c^2=-V^2\left(\frac{\partial P}{\partial V}\right)_S$$

$$PV^{\gamma}=const.$$

$$\left(\frac{\partial P}{\partial V}\right)_S=-\frac{\gamma P}{V}$$

$$\frac{P_2}{P_1}=\left(\frac{2}{\gamma+1}\right)^{\gamma/\gamma-1}$$

A high-velocity nozzle is designed to operate with steam at 700 kPa and 300°C. At the nozzle inlet the velocity is 30 m/s. Calculate values of the ratio A/A_1 (where A_1 is the cross-sectional area of the nozzle inlet) for the sections where the pressure is 600, 500, 400, 300, and 200 kPa. Assume the nozzle operates isentropically.

Initial values from the steam table: $S_1 = 7.2997 \frac{kJ}{kg \cdot K}$ $H_1 = 3059.8 \frac{kJ}{kg}$ $V_1 = 371.39 \frac{cm^3}{g}$

The continuity equation: $\frac{A}{A_1} = \frac{u_1 V}{V_1 u}$ → $\frac{A}{A_1} = \left(\frac{30}{371.39}\right)\frac{V}{u}$

Energy balance: $u^2 = u_1^2 - 2(H - H_1)$ → $u^2 = 900 - 2(H - 3059.8 \times 10^3)$

Since it is an isentropic process, $S = S_1$. From the steam table:

600 kPa: $S = 7.2997 \frac{kJ}{kg \cdot K}$ $H = 3020.4 \frac{kJ}{kg}$ $V = 418.25 \frac{cm^3}{g}$ → $u = 282.3 \frac{m}{s}$ $\frac{A}{A_1} = 0.120$

Similar for other pressures

P (kPa)	V (cm^3/g)	U (m/s)	A/A_1
700	371.39	30	1.0
600	418.25	282.3	0.120
500	481.26	411.2	0.095
400	571.23	523.0	0.088
300	711.93	633.0	0.091
200	970.04	752.2	0.104

Consider again the nozzle of the previous example, assuming now that steam behaves as an ideal gas. Calculate (a) the critical pressure ratio and the velocity at the throat. (b) the discharge pressure if a Mach number of 2.0 is required at the nozzle exhaust.

(a)

The ratio of specific heats for steam, $\gamma = 1.3$

$$\frac{P_2}{P_1} = \left(\frac{2}{\gamma+1}\right)^{\gamma/\gamma-1} \xrightarrow{\gamma = 1.3} \frac{P_2}{P_1} = 0.55$$

$$u_2^2 - u_1^2 = \frac{2\gamma P_1 V_1}{\gamma - 1}\left[1 - \left(\frac{P_2}{P_1}\right)^{\gamma-1/\gamma}\right]$$

We have u_1, P_1, V_1, P_2/P_1, γ → $u_2 = 544.35\ \frac{m}{s}$

(b)

$$\mathbf{M} = 2 \longrightarrow u_2 = 2 \times 544.35 = 1088.7\ \frac{m}{s}$$

$$u_2^2 - u_1^2 = \frac{2\gamma P_1 V_1}{\gamma - 1}\left[1 - \left(\frac{P_2}{P_1}\right)^{\gamma-1/\gamma}\right] \longrightarrow P_2 = 30.0\ kPa$$

Throttling Process

When a fluid flows through a restriction, such as an orifice, a partly closed valve, or a porous plug, without any appreciable change in kinetic or potential energy, the primary result of the process is a pressure drop in the fluid.

Throttling Valve

$$\frac{d(mU)_{cv}}{dt} + \Delta\left[\left(H + \frac{1}{2}u^2 + zg\right)\dot{m}\right]_{fs} = \dot{Q} + \dot{W}$$

$\dot{Q} = 0$

$\dot{W} = 0$

$$\Delta H = 0$$

Constant enthalpy

For ideal gas: $\Delta H = 0$ → $H_2 = H_1$ → $T_2 = T_1$

Throttling Process (continued)

For most real gas at moderate conditions of temperature and pressure, a reduction in pressure at constant enthalpy results in a decrease in temperature.

If a saturated liquid is throttled to a lower pressure, some of the liquid vaporizes or flashes, producing a mixture of saturated liquid and saturated vapor at the lower pressure. The large temperature drop results from evaporation of liquid. Throttling processes find frequent application in refrigeration.

Propane gas at 20 bar and 400 K is throttled in a steady-state flow process to 1 bar. Estimate the final temperature of the propane and its entropy change. Properties of propane can be found from suitable generalized correlations.

Constant enthalpy process:

$$\Delta H = \left\langle C_P^{ig} \right\rangle_H (T_2 - T_1) + H_2^R - H_1^R = 0$$

Final state at 1 bar: assumed to be ideal gas and $H_2^R = S_2^R = 0$

$$T_2 = \frac{H_1^R}{\left\langle C_P^{ig} \right\rangle_H} + T_1$$

$T_{r1} = 1.082$ $P_{r1} = 0.471$

And based on 2nd virial coefficients correlation

$$\left\langle C_P^{ig} \right\rangle_H = ??$$

$$\frac{H_1^R}{RT_c} = \left[\frac{(H_1^R)^0}{RT_c} + \omega \frac{(H_1^R)^1}{RT_c} \right] = HRB(TR, PR, OMEGA)$$

$$= HRB(1.082, 0.471, 0.152) = -0.452$$

$$C_P^{ig} = 1.213 + 28.785 \times 10^{-3} T - 8.824 \times 10^{-6} T^2$$

$T = 400K$ → $C_P^{ig} = 94.07 \dfrac{J}{mol \cdot K}$

???

$$T_2 = 385.2K$$

$$T = 0.5 \times 385.2 + 0.5 \times 400 = 392.6K$$

→ $\left\langle C_P^{ig} \right\rangle_H \approx C_P^{ig} = 92.73 \dfrac{J}{mol \cdot K}$

$$T_2 = 385.0K$$

$$\frac{S_1^R}{R} = SRB(1.082, 0.471, 0.152) = -0.2934$$

$$\Delta S = \left\langle C_P^{ig} \right\rangle_S \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} - S_1^R$$

$$\left\langle C_P^{ig} \right\rangle_S \approx \left\langle C_P^{ig} \right\rangle_H$$

$$\Delta S = 23.80 \frac{J}{mol \cdot K}$$

Throttling a real gas from conditions of moderate temperature and pressure usually results in a temperature decrease. Under what conditions would an increase in temperature be expected.

Define the **Joule/Thomson coefficient:** $\mu \equiv \left(\frac{\partial T}{\partial P}\right)_H$ → When will μ < 0 ???

$$\mu \equiv \left(\frac{\partial T}{\partial P}\right)_H = -\left(\frac{\partial T}{\partial H}\right)_P \left(\frac{\partial H}{\partial P}\right)_T = -\frac{1}{C_P}\left(\frac{\partial H}{\partial P}\right)_T$$

→ Sign of $\left(\frac{\partial H}{\partial P}\right)_T$???

Always negative

$$\left(\frac{\partial H}{\partial P}\right)_T = V - T\left(\frac{\partial V}{\partial T}\right)_P$$

$$V = \frac{ZRT}{P}$$

$$\left(\frac{\partial H}{\partial P}\right)_T = -\frac{RT^2}{P}\left(\frac{\partial Z}{\partial T}\right)_P$$

$$\mu = \frac{RT^2}{C_P P}\left(\frac{\partial Z}{\partial T}\right)_P$$

Always positive

$$\mu \leftrightarrow \left(\frac{\partial Z}{\partial T}\right)_P$$

Same sign

The condition $\left(\frac{\partial Z}{\partial T}\right)_P = 0$ may obtain locally for real gases. Such points define the Joule/Thomson inversion curve.

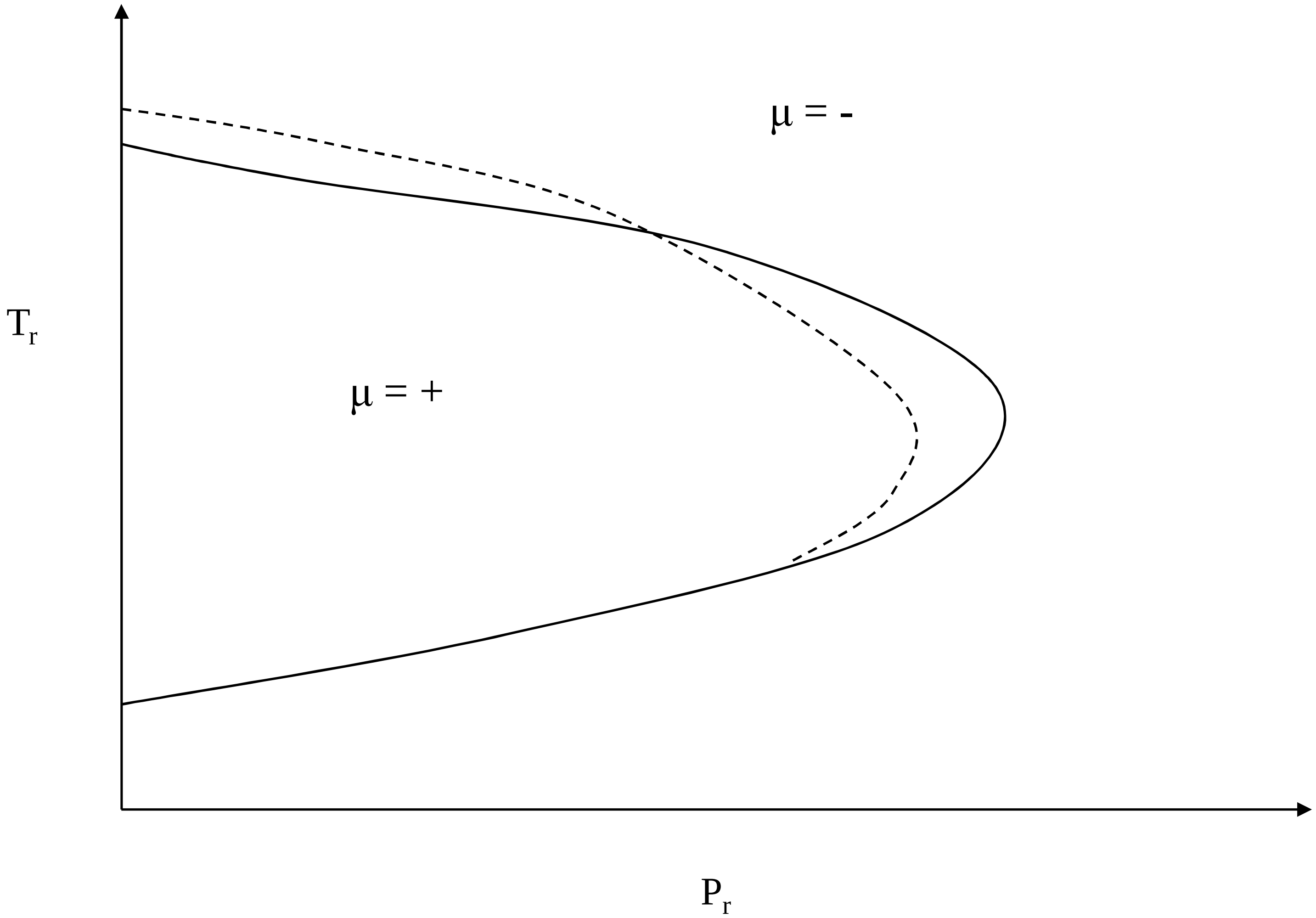
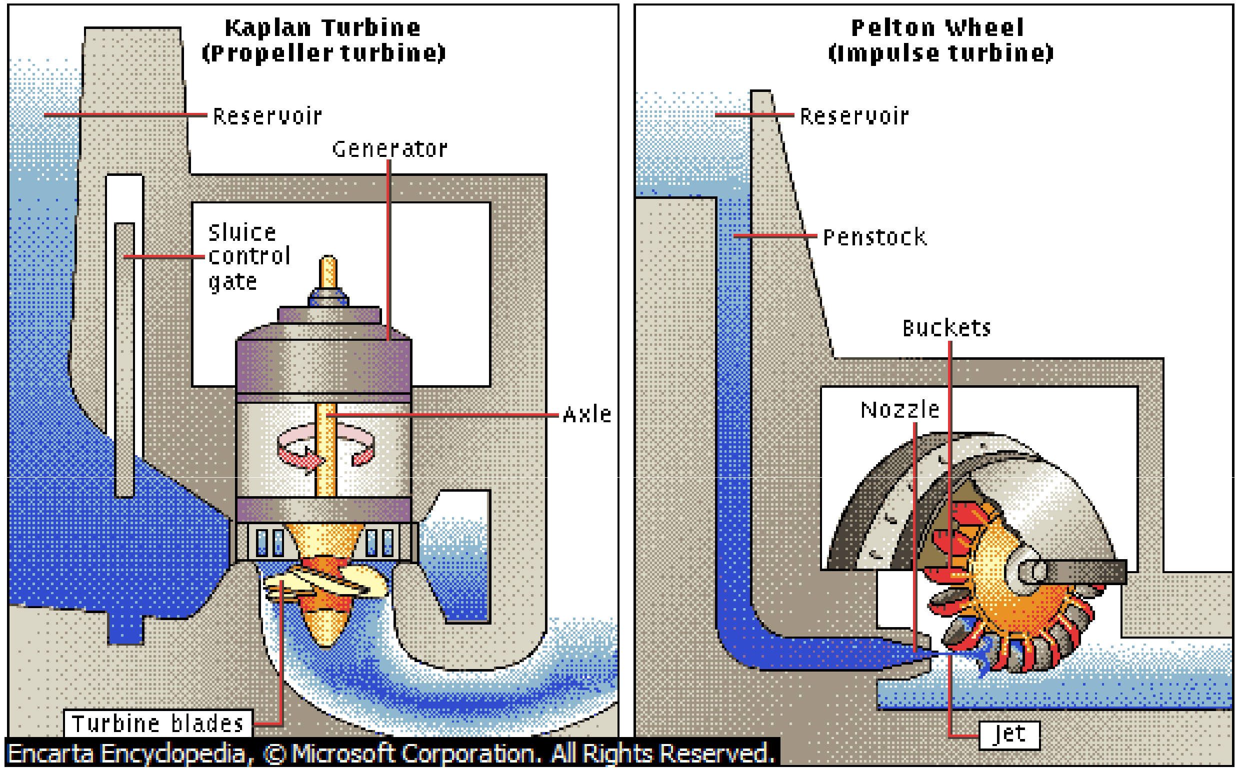


Fig 7.2

Turbine (Expanders)

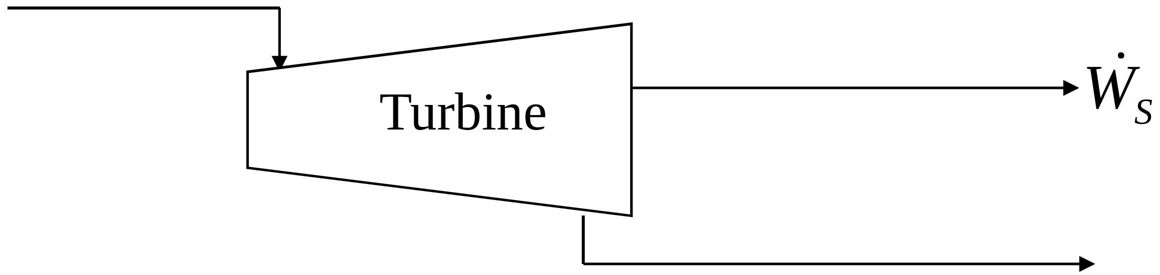
- A turbine (or expander):
  - Consists of alternate sets of nozzles and rotating blades
  - Vapor or gas flows in a steady-state expansion process and overall effect is the efficient conversion of the internal energy of a high-pressure stream into shaft work.

Turbin



Steam Turbine Blades from a
Calpine Geothermal Plant

DKB



$$\frac{d(mU)_{cv}}{dt} + \Delta\left[\left(H + \frac{1}{2}u^2 + zg\right)\dot{m}\right]_{fs} = \dot{Q} + \dot{W}_S$$

$$\dot{W}_S = \dot{m}\Delta H = \dot{m}(H_2 - H_1)$$

$$W_S = \Delta H = H_2 - H_1$$

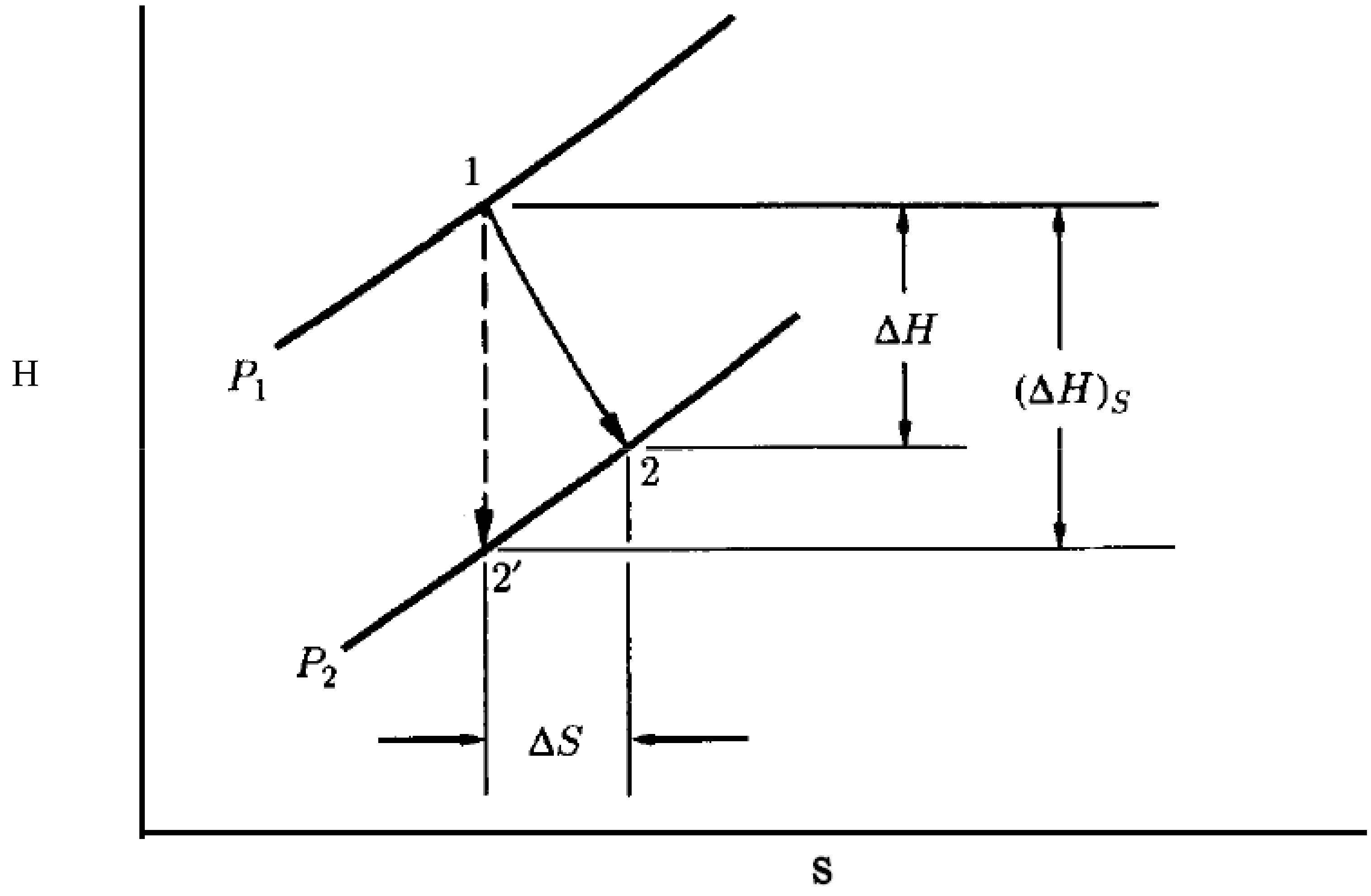
The maximum shaft work: a reversible process (i.e., isentropic, $S_1 = S_2$)

$$W_S(isentropic) = (\Delta H)_S$$

The turbine efficiency

$$\eta \equiv \frac{W_S}{W_S(isentropic)} = \frac{\Delta H}{(\Delta H)_S}$$

Values for properly designed turbines: 0.7~ 0.8

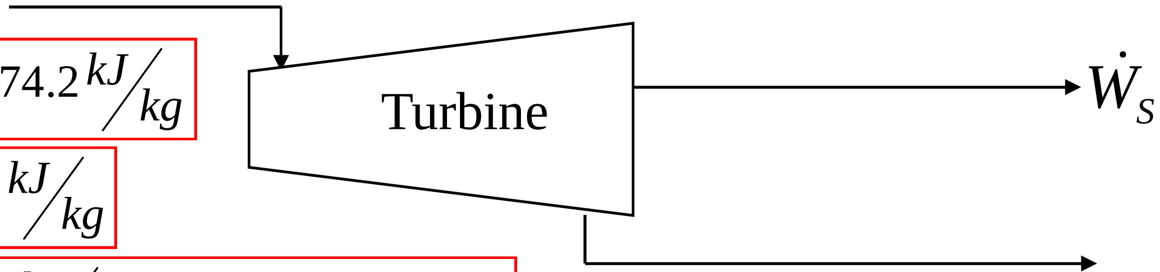


H
1
P_1
ΔH
$(\Delta H)_S$
2
2′
P_2
ΔS
S

A steam turbine with rated capacity of 56400 kW operates with steam at inlet conditions of 8600 kPa and 500°C, and discharge into a condenser at a pressure of 10 kPa. Assuming a turbine efficiency of 0.75, determine the state of the steam at discharge and the mass rate of flow of the steam.

$$P_1 = 8600\,kPa \quad T_1 = 500^\circ C$$

$$H_1 = 3391.6\,{}^{kJ}\!/_{kg} \quad S_1 = 6.6858\,{}^{kJ}\!/_{kg\cdot K}$$



$$P_2 = 10\,kPa \quad S_2' = 6.6858\,{}^{kJ}\!/_{kg\cdot K}$$

$$S_2' = (1-x^v)S_2^l + x^v S_2^v = (1-x^v)0.6493 + x^v 8.1511 = 6.6858\,{}^{kJ}\!/_{kg\cdot K}$$

$$x^v = 0.8047$$

$$H_2' = (1-x^v)H_2^l + x^v H_2^v = 2117.4\,{}^{kJ}\!/_{kg}$$

$$(\Delta H)_S = H_2' - H_1 = -1274.2\,{}^{kJ}\!/_{kg}$$

$$\Delta H = \eta(\Delta H)_S = -955.6\,{}^{kJ}\!/_{kg}$$

$$H_2 = H_1 + \Delta H = 2436.0\,{}^{kJ}\!/_{kg} = (1-x^v)H_2^l + x^v H_2^v$$

$$x^v = 0.9378$$

$$S_2 = (1-x^v)S_2^l + x^v S_2^v = 7.6846\,{}^{kJ}\!/_{kg\cdot K}$$

$$\dot{W}_S = \dot{m}\Delta H = 56400\,{}^{kJ}\!/_{s}$$

$$\dot{m} = 59.02\,{}^{kg}\!/_{s}$$

A stream of ethylene gas at 300°C and 45 bar is expanded adiabatically in a turbine to 2 bar. Calculate the isentropic work produced. Find the properties of ethylene by: (a) equations for an ideal gas (b)appropriate generalized correlations.

$P_1 = 45\ bar \quad P_2 = 2\ bar \quad T_1 = 573.15\ K$

$$\Delta H = \langle C_P^{ig} \rangle_H (T_2 - T_1) + H_2^R - H_1^R$$

$$\Delta S = \langle C_P^{ig} \rangle_S \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} + S_2^R - S_1^R$$

$$\Delta S = 0$$

(a) Ideal gas

$$\Delta S = \langle C_P^{ig} \rangle_S \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

$$\Delta S = 0$$

$$T_2 = \exp\left(\frac{-3.1135}{\langle C_P^{ig} \rangle_S / R} + 6.3511\right)$$

iteration

$$\langle C_P^{ig} \rangle_S / R = MCPS(573.15, T_2; 1.424, 14.394E-3, -4.392E-6, 0.0)$$

$$T_2 = 370.8K$$

$$W_S(isentropic) = (\Delta H)_S = \langle C_P^{ig} \rangle_H (T_2 - T_1)$$

$$\langle C_P^{ig} \rangle_H / R = MCPH(573.15, 370.18; 1.424, 14.394E-3, -4.392E-6, 0.0) = 7.224$$

$$W_S(isentropic) = 7.224 \times 8.314 \times (370.8 - 573.15) = -12153 \frac{J}{mol}$$

(b) General correlation

$T_{r1} = 2.030$ $P_{r1} = 0.893$

↓ based on 2nd virial coefficients correlation

$$\frac{H_1^R}{RT_c} = \left[\frac{(H_1^R)^0}{RT_c} + \omega \frac{(H_1^R)^1}{RT_c} \right] = HRB(2.030, 0.893, 0.087) = -0.234$$

$$\frac{S_1^R}{R} = SRB(2.030, 0.893, 0.087) = -0.097$$

Assuming $T_2 = 370.8$ K

$T_{r2} = 1.314$ $P_{r2} = 0.040$

↓ based on 2nd virial coefficients correlation

$$\frac{S_2^R}{R} = SRB(1.314, 0.040, 0.087) = -0.0139$$

iteration

$$\Delta S = \langle C_P^{ig} \rangle_S \ln \frac{T_2}{573.15} - R \ln \frac{2}{45} - 0.116 + 0.806 = 0$$

$$T_2 = 365.8K$$

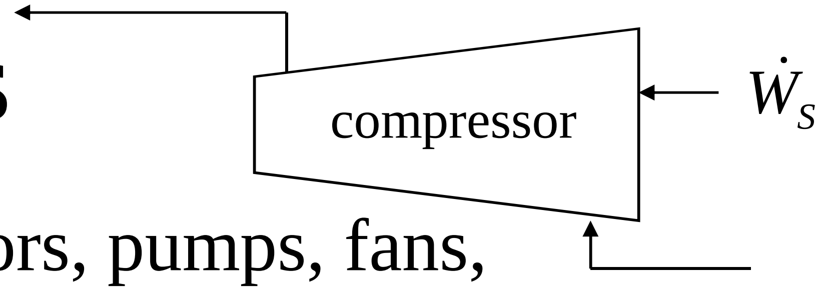
$T_{r2} = 1.296$ $P_{r2} = 0.040$

$$\frac{H_2^R}{RT_c} = HRB(1.296, 0.040, 0.087) = -0.20262$$

$$W_s(isentropic) = (\Delta H)_S$$

$$= \langle C_P^{ig} \rangle_H (T_2 - T_1) + H_2^R - H_1^R = -11920 \frac{J}{mol}$$

Compression process



- Pressure increases: compressors, pumps, fans, blowers, and vacuum pumps.
- Interested in the energy requirement

$$\frac{d(mU)_{cv}}{dt} + \Delta\left[\left(H + \frac{1}{2}u^2 + zg\right)\dot{m}\right]_{fs} = \dot{Q} + \dot{W}_S$$

$$\dot{W}_S = \dot{m}\Delta H = \dot{m}(H_2 - H_1)$$

$$W_S = \Delta H = H_2 - H_1$$

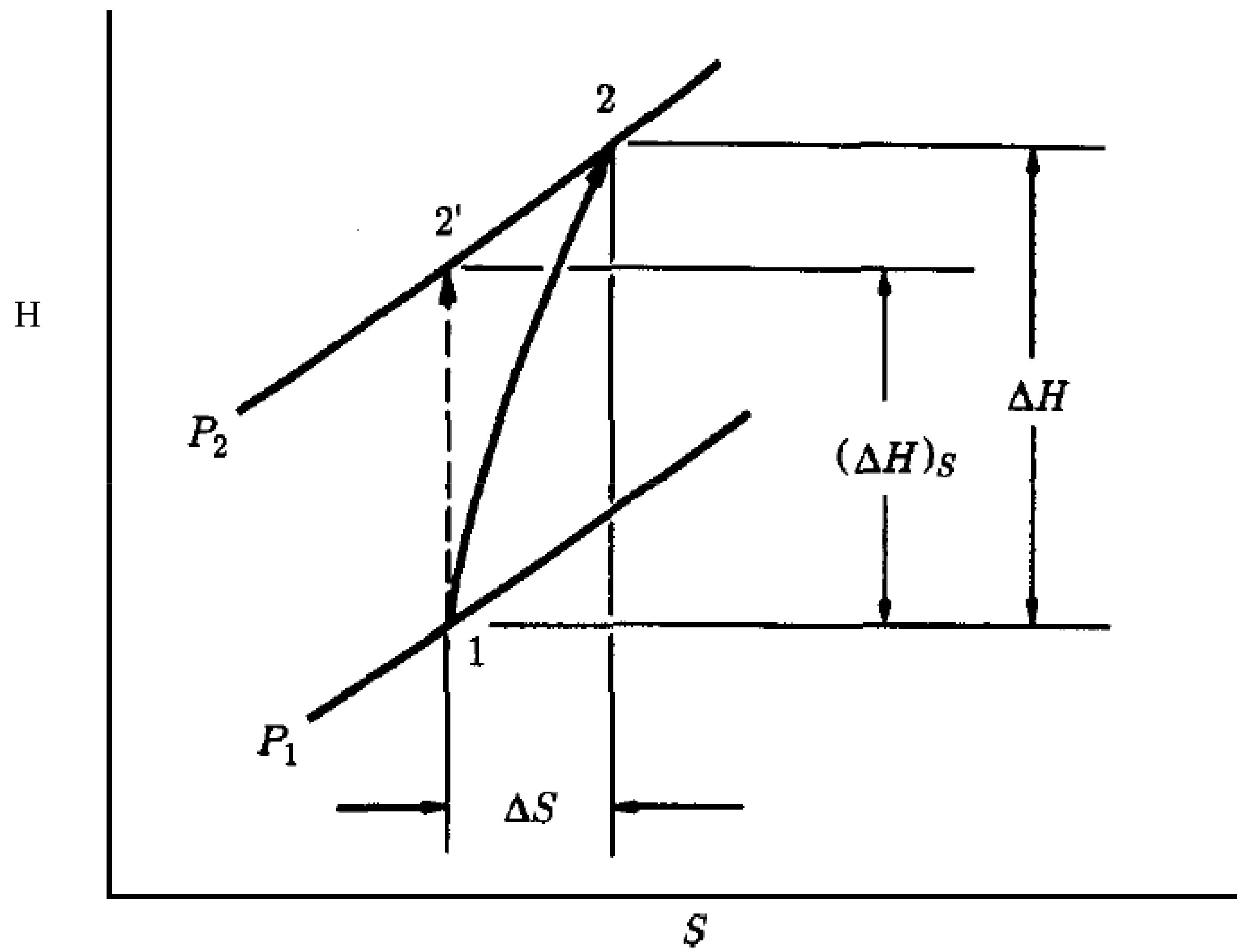
The minimum shaft work: a reversible process (i.e., isentropic, $S_1 = S_2$)

$$W_S(isentropic) = (\Delta H)_S$$

The compressor efficiency

$$\eta \equiv \frac{W_S(isentropic)}{W_S} = \frac{(\Delta H)_S}{\Delta H}$$

Values for properly designed compressors: 0.7~ 0.8



H
2
2'
P2
(ΔH)S
ΔH
1
P1
ΔS
S

Saturated-vapor steam at 100 kPa (t^{sat} = 99.63 °C) is compressed adiabatically to 300 kPa. If the compressor efficiency is 0.75, what is the work required and what is the work required and what are the properties of the discharge stream?

For saturated steam at 100 kPa: $S_1 = 7.3598 \frac{kJ}{kg \cdot K}$ $H_1 = 2675.4 \frac{kJ}{kg}$

Isentropic compression

$S_2' = S_1 = 7.3598 \frac{kJ}{kg \cdot K}$ → 300 kPa → $H_2' = 2888.8 \frac{kJ}{kg}$ → $(\Delta H)_S = 213.4 \frac{kJ}{kg}$

$$\Delta H = \frac{(\Delta H)_S}{\eta} = 284.5 \frac{kJ}{kg}$$

$$W_S = \Delta H = 284.5 \frac{kJ}{kg}$$

$$H_2 = H_1 + \Delta H = 2959.9 \frac{kJ}{kg}$$

300 kPa →

$T_2 = 246.1^\circ C$

$S_2 = 7.5019 \frac{kJ}{kg \cdot K}$

If methane (assumed to be an ideal gas) is compressed adiabatically from 20°C and 140 kPa to 560 kPa, estimate the work requirement and the discharge temperature of the methane. The compressor efficiency is 0.75.

$$\Delta S = \left\langle C_P^{ig} \right\rangle_S \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} + S_2^R - S_1^R$$

$$\Delta S = 0$$

$$\left\langle C_P^{ig} \right\rangle_S \Big/ R = MCPS(293.15, T_2; 1.702, 9.081E-3, 2.164E-6, 0.0)$$

$$T_2 = T_1 \left(\frac{P_2}{P_2} \right)^{R \big/ \left\langle C_P^{ig} \right\rangle_S}$$

$$\left\langle C_P^{ig} \right\rangle_H \Big/ R = MCPH(293.15, T_2; 1.702, 9.081E-3, 2.164E-6, 0.0)$$

iteration

$$P_2 / P_1 = 4 \qquad T_1 = 293.15K$$

$$T_2 = 397.37K$$

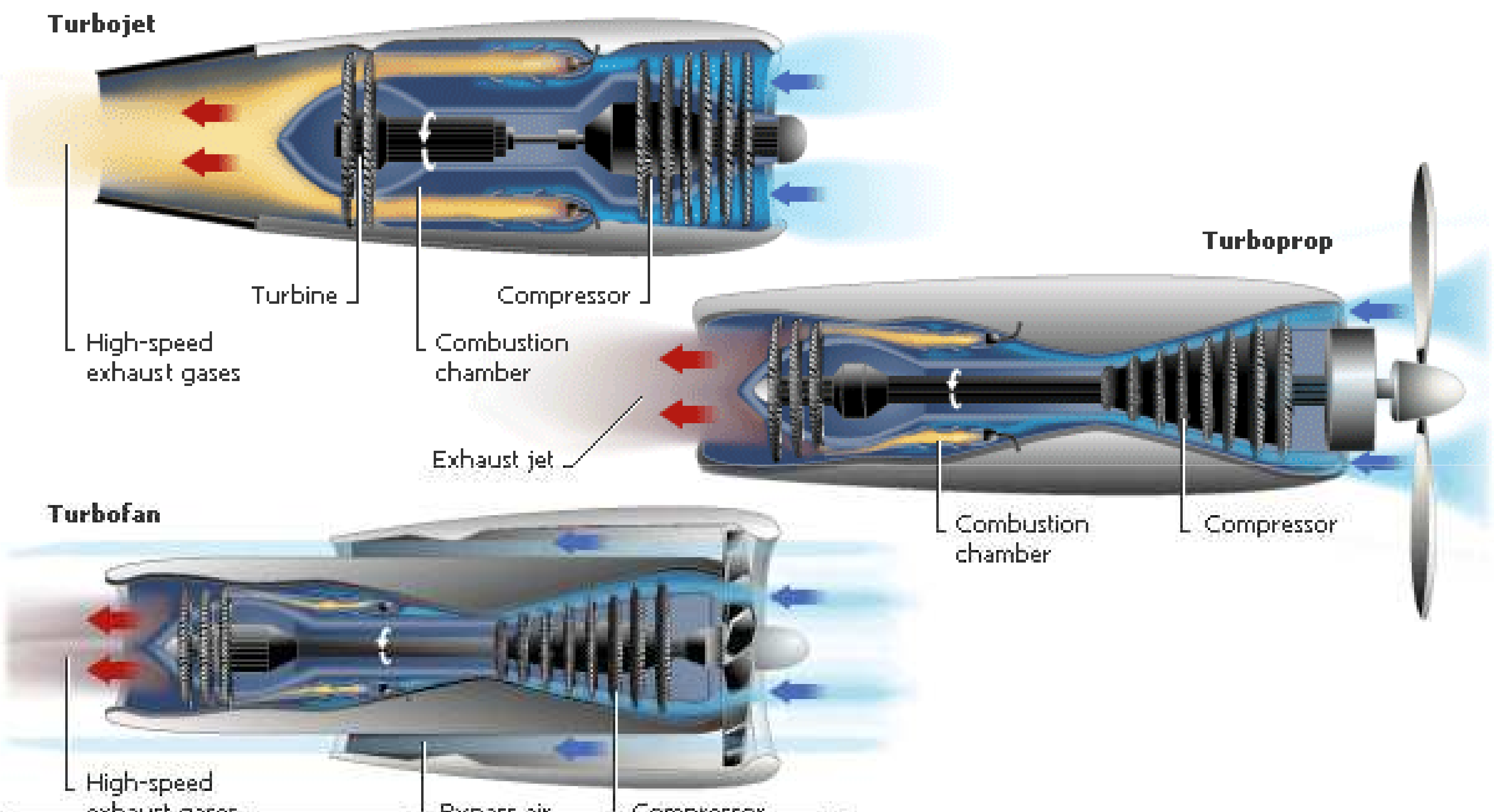
$$W_s(isentropic) = (\Delta H)_S = \left\langle C_P^{ig} \right\rangle_H (T_2 - T_1) + H_2^R - H_1^R$$

$$W_s(isentropic) = 3966.2 \frac{J}{mol}$$

$$W_s = \frac{W_s(isentropic)}{\eta} = 5288.3 \frac{J}{mol}$$

$$W_s = \Delta H = \left\langle C_P^{ig} \right\rangle_H (T_2 - T_1)$$

$$T_2 = 428.65K$$


Turbojet
Turbine
Compressor
High-speed
exhaust gases
Combustion
chamber
Exhaust jet
Turboprop
Combustion
chamber
Compressor
Turbofan
High-speed

Pumps

- Liquids are usually moved by pumps. The same equations apply to adiabatic pumps as to adiabatic compressors.
- For an isentropic process: $W_s(isentropic) = (\Delta H)_S = \int_{P_1}^{P_2} VdP$
- With $dH = C_P dT + V(1-\beta T)dP$ $\quad dS = C_P \dfrac{dT}{T} - \beta VdP$
- For liquid,
  - $W_s(isentropic) = (\Delta H)_S = V(P_2 - P_1)$
  - $\Delta H = C_P \Delta T + V(1-\beta T)\Delta P$
  - $\Delta S = C_P \ln \dfrac{T_2}{T_1} - \beta V \Delta P$

Water at 45°C and 10 kPa enters an adiabatic pump and is discharged at a pressure of 8600 kPa. Assume the pump efficiency to be 0.75. Calculate the work of the pump, the temperature change of the water, and the entropy change of water.

The saturated liquid water at 45°C: $V = 1010 \dfrac{cm^3}{kg}$ $\beta = 425 \times 10^{-6} \dfrac{1}{K}$ $C_P = 4.178 \dfrac{kJ}{kg \cdot K}$

$$W_s(isentropic) = (\Delta H)_S = V(P_2 - P_1)$$

$$W_s(isentropic) = 1010 \times (8600 - 10) = 8.676 \times 10^6 \frac{kPa\, cm^3}{kg} = 8.676 \frac{kJ}{kg}$$

$$W_s = \frac{W_s(isentropic)}{\eta} = \Delta H = 11.57 \frac{kJ}{kg}$$

$$\Delta H = C_P \Delta T + V(1 - \beta T)\Delta P$$

$$\Delta T = 0.97 K$$

$$\Delta S = C_P \ln \frac{T_2}{T_1} - \beta V \Delta P$$

$$\Delta S = 0.0090 \frac{kJ}{kg \cdot K}$$